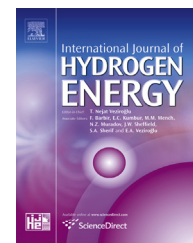


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Optimization of molten carbonate fuel cell (MCFC) and homogeneous charge compression ignition (HCCI) engine hybrid system for distributed power generation[☆]

Seonyeob Kim^a, Ji Young Jung^a, Han Ho Song^{a,*}, Seung Jin Song^a,
Kook Young Ahn^b, Sang Min Lee^b, Young Duk Lee^b, Sanggyu Kang^b

^a Department of Mechanical & Aerospace Engineering, Seoul National University, Gwanak-gu, Seoul, South Korea

^b Korea Institute of Machinery and Materials, 104 Sinseongno, Yuseong-Gu, Daejeon, South Korea

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ABSTRACT

In a previous study, a new hybrid system of molten carbonate fuel cell (MCFC) and homogeneous charge compression ignition (HCCI) engine was developed, where the HCCI engine replaces the catalytic burner and produces additional power by using the left-over heating values from the fuel cell stack. In the present study, to reduce the additional cost and footprint of the engine system in a hybrid configuration, the possibility of engine downsizing is investigated by using two strategies, i.e. the use of a turbocharger and the use of high geometric compression ratio for the engine design, both of which are to increase the density of the intake charge and thus the volumetric efficiency of the engine. Combining these two strategies, we suggest a new engine design with ~60% of displacement volume of the original engine. In addition, operating strategies are developed to run the new hybrid system under part load conditions. It is successfully demonstrated that the system can operate down to 65% of the power level of the design point, while the system efficiency remains almost unchanged near 63%.

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1. Introduction

Recently, interest in renewable resources and energy conversion systems has increased due to environmental concerns and conventional fossil fuel depletions. Fuel cell technology is considered as a promising candidate for this category with its high efficiency and almost zero pollutant emission. Among

various types of fuel cells, those operating at high temperatures (~600–800 °C) generally show higher efficiency than those at relatively lower temperatures [1]. There are two major types of high-temperature fuel cells, which are solid oxide fuel cell (SOFC) and MCFC. The former is still under active research and development stage, but the latter is already commercialized for distributed power generation purpose.

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* Corresponding author. Tel./fax: +82 2 880 1651.

E-mail address: hhsong@snu.ac.kr (H.H. Song).

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Nomenclature			
A_{seg}	area of each segment (cm^2)	R_{total}	total irreversible loss ($\Omega \text{ cm}^{-2}$)
CI	compression ignition	R	specific heat ratio
ΔF	change in molar flow rate (mol h^{-1})	T	temperature (K)
ΔG	gibbs free energy change (J mol^{-1})	V_{opn}	operating cell voltage (V)
ΔH	apparent activation energy for resistance parameters (J mol^{-1})	W	work output (J)
E_{rev}	maximum reversible potential (V)	X	conversion degree
f	faraday's constant (96487 C mol^{-1})	Greek letter	
F	molar flow rate (mol h^{-1})	η	loss
H	frequency factor for resistance parameters (J mol^{-1})	Subscripts	
i	current (A)	and	anode
j	current density (A cm^{-2})	and, i	species at anode
K_p	equilibrium constant	cat	cathode
M	molar fraction	cat, i	species at cathode
NO_x	oxides of nitrogen	CO	carbon monoxide
ODEs	ordinary differential equations	CO_2	carbon dioxide
P	partial pressure (bar)	H_2	hydrogen molecule
Q	energy required (J)	H_2O	water
R	universal gas constant ($\text{J mol}^{-1} \text{ K}^{-1}$)	i	species
R_{and}	irreversible loss at anode ($\Omega \text{ cm}^{-2}$)	Nern	Nernst
R_{cat}	irreversible loss at cathode ($\Omega \text{ cm}^{-2}$)	X	gas
R_{int}	internal cell resistance ($\Omega \text{ cm}^{-2}$)	c	compressor
		t	turbine
		comb	combustion

Several studies have attempted to hybridize these high-temperature fuel cell systems with other power generating technologies, in order to achieve additional power and efficiency of the overall system. Especially, due to its operating characteristics of MCFC, there have been many efforts to build the hybrid system on it. First of all, hybrid system of MCFC coupled with gas turbine (GT) was studied by several researchers [2–7]. Lunghi et al. analyzed and optimized an MCFC and GT hybrid system, which was shown to achieve over 58% efficiency by parametric performance evaluation [2]. Roberts et al. performed a dynamic simulation of an MCFC and GT hybrid system, and in their analysis, the hybrid system efficiency was ~57% [3]. Ubertini et al. proposed an MCFC system combined with a steam injected GT. The overall system efficiency was 69% [4]. Liu et al. presented a hybrid system composed of a pressurized MCFC and a micro GT. They simulated the hybrid system at both design and off-design operations, and the system efficiency was ~58% [5]. Rashidi et al. developed an MCFC system with a turbo expander, which had the system efficiency of ~60% [6]. Orecchini et al. analyzed an MCFC and micro-turbine hybrid system by using an in-house MCFC model with a plate reformer, and micro turbine-compressor model. The electrical efficiency of their hybrid system was ~74% [7]. Secondly, organic Rankine cycle (ORC) was proposed as a bottoming cycle of the MCFC for waste heat recovery [8–10]. Angelino et al. studied an MCFC plant with ORCs using various working fluids. The system efficiency was ~60% [8]. Sanchez et al. introduced a hybrid system, which is composed of MCFC, GT, and ORC. They performed an optimization of the system, and the hybrid system efficiency was ~59% [9]. Recently, Vatani et al. analyzed the hybrid system of ORC and a direct internal reforming MCFC (DIR-MCFC), which showed an efficiency of

~60% [10]. Finally, a hybrid system of MCFC with Stirling engine was also suggested [11,12]. Sanchez et al. developed an MCFC-Stirling engine hybrid system, and compared it with MCFC-Rankine and MCFC-Brayton systems. The efficiency of the former was ~61%, which was higher than the efficiencies of the other systems [11]. Recently, Escalona et al. analyzed two hybrid systems, which were MCFC-Stirling system and MCFC-SCO₂ (supercritical carbon dioxide turbine) system. The MCFC-Stirling system efficiency was 56% and the MCFC-SCO₂ system efficiency was ~59% [12]. Although many hybrid systems were introduced, most researches were focused on large-scale power generation, except for MCFC-Stirling system.

Recently, we developed a new MCFC-HCCI engine hybrid system for sub-megawatts distributed power generation [13]. This new hybrid system was modeled by using the Mathworks Matlab and Cantera toolbox, and compared with the stand-alone counterpart. At comparable boundary conditions, the former demonstrated ~20% (relative) improvement in both power output and system efficiency, as compared to the latter, which proved itself to be a promising candidate for highly-efficient distributed power generation system in the near future. Fig. 1 shows a schematic of (a) standalone system using catalytic burner and (b) hybrid system using HCCI engine.¹

In this study, we present the results on the feasibility of the hybrid system in a real-world application. There were three major issues discussed in our previous work [13]:

1. To downsize the HCCI engine in the hybrid system

¹ Turbocharger, shown in Fig. 1, was not included in our previous hybrid system.

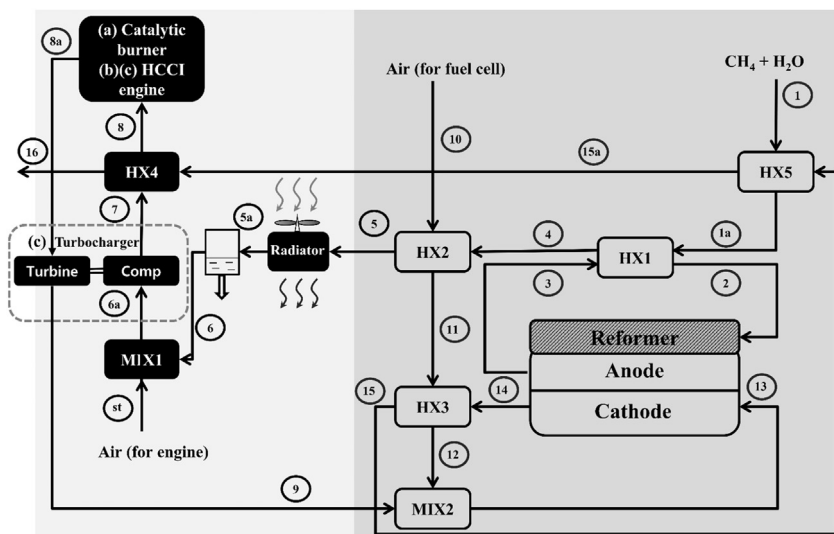


Fig. 1 – System configurations: (a) standalone system with catalytic burner, (b) hybrid system with HCCI engine, and (c) new hybrid system with HCCI engine and turbocharger.

2. To demonstrate part-load or off-design operation of the hybrid system
3. To develop start-up or shut-down strategies of the hybrid system

We are focusing on the first two topics in this paper, and the third topic is still under investigation.

In the first topic, our goal is to reduce the required HCCI engine size and thus increase the specific power output (or kW/liter). Our previous system adopted an 18.4-L HCCI engine for the hybridization with a 260-kW MCFC system, and the engine output at design point operation was ~55 kW. It is noted that the typical power output in the comparable size of the engine is ~10 times higher, or ~500 kW, which implies that the engine size in the original design is not at all optimized and may cause significant amount of extra cost and system volume that may be avoided otherwise. Therefore, downsizing the engine is a key factor to improve the feasibility of the hybrid system.

The required engine size in the hybrid system is directly correlated with the intake volume flow rate to the engine (state 8), which is a mixture of the reduced anode off-gas (state 6) and the air (state st). If the engine tries to intake more volume than supplied, it may create the vacuum pressure in the system; on the other hand, if the engine can't take the supplied volume flow rate, it can result in back-pressure in the system. Thus, to achieve the engine downsizing, the intake volume flow rate to the engine should be decreased.²

To decrease the intake volume flow rate, one may consider the following options: 1. mass flow rate reduction to the engine, 2. higher intake pressure, and 3. lower intake temperature. As mentioned earlier, the engine intake charge consists

of the anode off-gas from the MCFC and the fresh air as an oxidizer for the engine operation. Here, the anode off-gas is composed of CO, CO₂, H₂ and H₂O. Among these species, H₂O, ~30% of the total mass, is relatively easy to be removed by condensation to reduce the mass flow rate to the engine. In our original system, the water-condensing radiator for the anode off-gas was already adopted, as shown in Fig. 1. In this study, we focus on the two other options: to achieve the higher intake pressure and the lower intake temperature, the turbocharger and the engine with higher geometric compression ratio are considered, respectively.

In the second topic of part-load operation, our goal is to develop operating strategies for power variation. It is noted that the fuel cell hybrid system should be able to run at various power levels, although it is generally operated at a constant or designed power output level. Due to the complication of the additional degree of freedom by having the engine in the hybrid system, new strategies need to be developed. Since the commercialized MCFC standalone system already has well-developed off-design operation capability, our focus is more on the possible issues with the HCCI engine in the hybrid system. These issues include: 1. volume flow rate control with the reduced mass flow rate from the fuel cell at part-load operation, and 2. HCCI combustion timing control to ensure proper engine operation.

The main body of this paper consists of three sections: Modeling Descriptions, Engine Downsizing, and Off-design Operation. In Modeling Descriptions section (Section 2), the new hybrid system design with the turbocharger is explained, followed by detailed modeling descriptions on the major components, i.e. MCFC stack, HCCI engine, and turbocharger. In Engine Downsizing section (Section 3), the aforementioned techniques for downsizing the engine, i.e. the use of turbocharger and high geometric compression ratio, are discussed. Finally, in Off-design Operation section (Section 4), the newly-developed system with the downsized engine from Section 3 is used to develop the operating strategies of the hybrid

² The engine speed can be increased to accommodate higher volume flow rate in a given engine size, but in this study, we fix the speed at 1800 rpm, which is a typical number for the engine sizes that we consider.

system at various power levels. The system performance at off-design operation will also be provided.

2. Modeling descriptions

In this section, detailed modeling descriptions on the hybrid system and its components are provided. These components include MCFC stack, HCCI engine, heat exchangers, mixers, and turbocharger. The modeling was performed by using the Mathworks MATLAB and Cantera toolbox which provides functions regarding thermodynamic properties and chemical reaction kinetics. It is noted that most of the component models were previously developed by the authors except the turbocharger model, and, therefore, only the essentials are reproduced in this section. The readers are referred to Ref. [13] for further details.

Here is the brief overview of this section. Firstly, new hybrid system configuration with the turbocharger is discussed for various gas flows and several modeling assumptions. Secondly, one-dimensional, non-isothermal MCFC stack model is described for internal flows, various losses, and temperature change along the channels. Thirdly, zero-dimensional HCCI engine model is introduced with engine specifications and operating conditions. Finally, the details on the turbocharger model newly developed in this study are presented.

2.1. Hybrid system

New hybrid system configuration with the turbocharger is shown in Fig. 1, as compared to the standalone system and the previous hybrid system without turbocharger. There are two additional states, i.e. states 6a and 8a for the compressor inlet and the turbine inlet of the turbocharger, respectively.

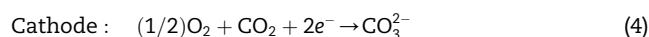
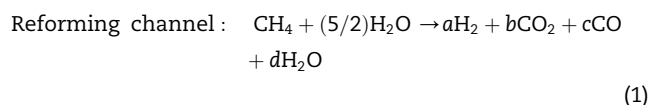
The working principles are described in the followings. Methane (CH_4) with steam (H_2O) (state 1) enters the system with the mole ratio of 1:2.5. This fuel gas is preheated at HX5 by cathode off-gas (state 15) and at HX1 by anode off-gas (state 3) to achieve the MCFC operating temperature. The air for MCFC operation (state 10) is preheated at HX2 and HX3, where it exchanges heats with anode off-gas (state 4) and cathode off-gas (state 14), respectively. After preheated, the air is mixed with engine exhaust gas (state 9) in MIX2, and then enters the cathode of MCFC. The anode off-gas (state 5), after HX1 and HX2, enters the radiator, where it is cooled down to condense out water by considering the saturation pressure. Then, the reduced anode off-gas (state 6) is mixed with the air for the engine operation (state st) in MIX1. This mixture (state 6a) enters the compressor, and then heated at HX4 by the cathode off-gas (state 16) which exits the system afterward. Typically, post-compressor temperature (state 7) is already higher than the ambient temperature, but the HCCI operation requires even higher temperature (state 8), which is achieved by controlling the degree of heat exchange at HX4. After engine operation, the exhaust gas (state 8a) comes in the turbine. After expanded to the atmospheric pressure through the turbine, the exhaust gas (state 9) mixes with the cathode air at MIX2. In our analysis, all the heat exchangers have counter-

flowing configuration, and their effectiveness is kept under 85% for more realistic operation. For the mixers, adiabatic mixing process is assumed.

2.2. MCFC stack modeling

One-dimensional, non-isothermal, MCFC stack model was previously developed by the authors, to analyze the performance of the fuel cell in the hybrid system [13]. The model took a similar approach as described in Ref. [14] by Baranak et al., although we newly added reforming channel and non-isothermal fuel cell operations in the model. In this section, major assumptions, features, and flow chart for the stack model are presented, and the readers are referred to the aforementioned references for the detailed descriptions on equations, parameters, and assumptions.

The fuel mixture of methane and steam enters the reforming channel first and then the anode channel. In the reforming channel, methane-steam mixture is transformed into the mixture of H_2 , CO_2 , CO , and H_2O , whose composition is determined by atom balances and water–gas-shift-reaction (WGSR). The mixture of the cathode air and engine exhaust gas runs into the cathode channel, where oxygen molecule (O_2) and carbon dioxide (CO_2) form carbonate ion (CO_3^{2-}) with the electrons from the anode side. The CO_3^{2-} ion is transferred through the electrolyte to the anode side as an oxygen carrier. Then, hydrogen molecule (H_2) in the anode side reacts with the oxygen in CO_3^{2-} ion to produce the electrons and product water (H_2O). The followings are the various reactions considered in the fuel cell modeling.



The major assumptions for the MCFC stack model are:

- Cathode and anode streams flow in parallel and assume the same temperature changes along the channels.
- Pressure change across the fuel cell channel is negligible.
- In reforming channel, the fuel mixture is fully reformed and achieves thermodynamic equilibrium at the exit, before it enters the anode channel.
- The water–gas-shift-reaction at anode channel occurs fast enough to keep anode compositions at equilibrium all the time.
- Overall system operates under steady state condition.
- Natural gas, a fuel for typical MCFC system, is modeled as pure methane.

To consider the effects of varying thermodynamic states along the channels, i.e. temperature and composition changes, on the fuel-cell performance, the fuel cell is divided into 25 segments in the direction of anode and cathode flows.

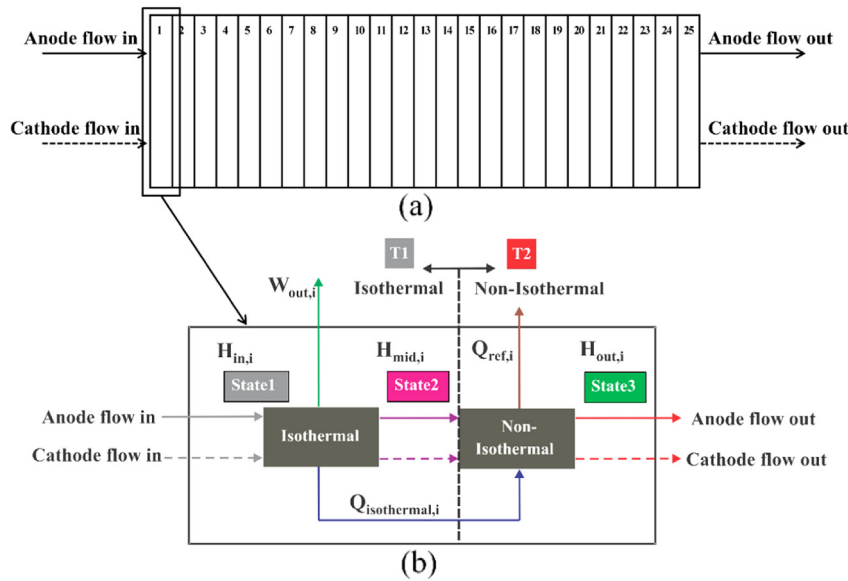


Fig. 2 – MCFC stack modeling schematic: (a) parallel flows with 25 segments and (b) energy flows within each segment.

Fig. 2 schematically shows (a) the 25 segments and (b) energy and mass flows at each segment which consists of an isothermal and a non-isothermal part. At the isothermal part, electrochemical reactions occur at constant temperature of the inlet temperature of the segment (state 1 in Fig. 2), and the current density is determined based on the operating cell voltage and the calculated irreversible losses. Then, the WGSR equilibrates the composition of anode gas mixture. Based on these calculations, the work produced by the designated segment ($W_{out,i}$) can be evaluated, as well as the quantity of the heat transfer $Q_{isothermal,i}$, which is to keep the constant temperature in the isothermal part. It is noted that the overall electrochemical reaction in the fuel-cell operation is oxidation reaction or net-exothermic, and thus $Q_{isothermal,i}$ typically takes positive values. At the non-isothermal part, the outlet temperature of the segment is calculated by considering the exothermicity $Q_{isothermal,i}$ from the isothermal part, and the heat transfer $Q_{ref,i}$ which is required by the reforming reaction in the reforming channel.

Fig. 3 shows the flow chart for the stack modeling process. In the beginning, the fuel cell operating voltage is arbitrarily assumed. With this value, the calculation is performed for the isothermal and the non-isothermal parts of the first segment. At the isothermal part, current density and work output are calculated on the basis of the operating voltage, as well as the composition changes by considering electrochemical reactions and WGSR. At the non-isothermal part, the first law of thermodynamics is applied to calculate the outlet temperature. The outlet conditions of the first segment become the inlet conditions of the second segment, and so on. After all the calculations are finished for 25 segments, the total current and work output are determined and the fuel utilization factor is calculated by using the lower heating values of fuel mixture (state 2 in Fig. 1) and anode off-gas (state 3 in Fig. 1). If the utilization factor doesn't match with the desired value (0.7 in this study), the whole processes are iterated with the newly assumed operating voltage.

2.3. HCCI engine modeling

The HCCI engine model was previously developed and validated by the authors [15,16]. The model uses the Mathworks MATLAB with Cantera toolbox, which allows the

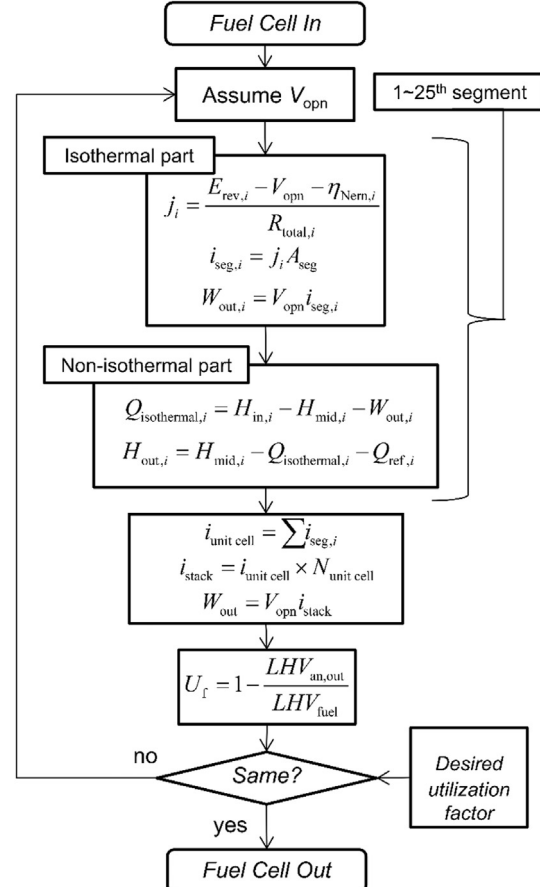


Fig. 3 – MCFC stack modeling flow chart.

thermodynamic properties and chemical reaction kinetics calculation. In a typical HCCI operation, well-mixed reactant mixture is introduced during the intake process. During the main compression stroke, the reactant mixture is compressed up to the point where the volumetric self-ignition occurs throughout the cylinder, without any external ignition source, such as, a spark-plug in spark-ignition engine (a.k.a. gasoline engine) or a direct-fuel-injection-system in compression-ignition engine (a.k.a. diesel engine). Therefore, the combustion timings are largely dependent on the thermodynamic states of the reactants and chemical reaction kinetics. In typical HCCI operation, the temperature and composition of the intake mixture become the major control variables in combustion timings.

In HCCI engine, combustion timing is one of the most determining factors in efficiency. Firstly, too early combustion timing, near or before the top dead center (TDC), has a negative effect on the efficiency by increased heat transfer, because high-temperature post-combustion gas is exposed to the relatively cold cylinder wall with the highest surface-to-volume ratio near TDC. Secondly, too late combustion timing may also degrade the cycle efficiency by possible incomplete combustion or even misfire. As the piston moves down after the top dead center, the mixture expands and cools down to the level when the mixture is no more reactive enough to complete the combustion reactions. Therefore, the proper combustion timing should be maintained to maximize the efficiency of the HCCI engine.

Due to the nearly homogenous nature of the mixture preparation and combustion phenomena in HCCI engine, our engine model assumes a homogeneous thermodynamic state for the in-cylinder mixture during the closed parts of the engine cycle, i.e. compression and expansion strokes. In order to predict the combustion timing, GRI-Mech 3.0, containing 53 species and 325 reactions, is adopted, which was developed for natural gas application [17]. The heat transfer to the cylinder wall is evaluated by using the modified Woschni model [18]. The detailed modeling descriptions can be found in Appendix.

In our modeling, the intake and exhaust processes are treated separately to calculate the pumping loss, which is caused by the difference in the pressures during the intake and exhaust strokes. Since there is no intake throttling applied in typical HCCI operation, both intake and exhaust manifolds are assumed to be at atmospheric pressures, and thus the pumping loss is usually zero. However, in our new engine design, we are applying the turbocharger, which may cause the pressure difference in the intake and exhaust manifolds. In this case, the pumping loss is evaluated by multiplying this difference in pressures by the engine displacement volume.

Finally, the combustion efficiency is evaluated to determine the degree of completeness of the combustion and defined as in Eqn. (5).

$$\eta_c = 1 - \frac{\text{LHV}_{\text{out}}}{\text{LHV}_{\text{in}}} \quad (5)$$

where LHV_{in} is the lower heating value of the fuel supplied, and LHV_{out} is the lower heating value of the exhaust gas. In this study, the combustion efficiency over 98% is assumed to be complete combustion.

2.4. Turbocharger modeling

A turbocharger consists of compressor and turbine, interconnected by a shaft. The turbine exploits the enthalpy left in the exhaust gas of the engine to produce the shaft work, which is delivered to operate the compressor. In a given engine size, the pressurized intake charge through the compressor leads to increased mass flow rate and thus work output per engine cycle. On the other hand, in our application, the mass flow rate to the engine is already given at certain operating condition of the fuel cell, and thus the use of the compressor to pressurize the intake charge can lead to reduced volume flow rate and smaller engine size requirement.

To examine the possibility of the engine downsizing by using a turbocharger, a simple thermodynamic turbocharger model is developed. We referred to Ref. [19] by Moraal et al., and the main governing equations are reproduced here with some modifications to the original model. Table 1 lists the major parameters used in the modeling.

The pressure ratio (PR) of the compressor, which is the ratio between the inlet and outlet pressures, is the most important parameter in the turbocharger operation. Once the PR is given, the pressure of the gas discharged from the compressor is calculated as in Eqn. (6).

$$P_{c,\text{out}} = \text{PR} \times P_{c,\text{in}} \quad (6)$$

where $P_{c,\text{out}}$ is the discharged gas pressure from the compressor, and $P_{c,\text{in}}$ is the inlet gas pressure. Then, the required power by the compressor is given by Eqn. (7).

$$\dot{W}_c = \dot{m}_c c_{p,c} (T_{c,\text{out}} - T_{c,\text{in}}) \quad (7)$$

where $\dot{m}_c$ is the mass flow rate through the compressor, $c_{p,c}$ is the specific heat of the gas, and $T_{c,\text{in}}$ and $T_{c,\text{out}}$ are the temperatures at the compressor inlet and outlet, respectively.

If the compressor is working in an isentropic process, the relation between the temperature and the pressure is given in Eqn. (8).

$$\frac{T_{c,\text{out},s}}{T_{c,\text{in}}} = \left(\frac{P_{c,\text{out}}}{P_{c,\text{in}}} \right)^{\gamma_c - 1/\gamma_c} \quad (8)$$

where $T_{c,\text{out},s}$ is the outlet temperature when isentropically compressed, and γ_c is the specific heat ratio.

Then, by assuming constant specific heats, isentropic efficiency is defined as in Eqn. (9), and thus the outlet temperature of the non-isentropic compressor is calculated as in Eqn. (10).

$$\eta_{c,s} = \frac{T_{c,\text{out},s} - T_{c,\text{in}}}{T_{c,\text{out}} - T_{c,\text{in}}} \quad (9)$$

$$\frac{T_{c,\text{out}}}{T_{c,\text{in}}} = 1 + \frac{\left((P_{c,\text{out}}/P_{c,\text{in}})^{\gamma_c - 1/\gamma_c} - 1 \right)}{\eta_{c,s}} \quad (10)$$

where $\eta_{c,s}$ is the isentropic efficiency of the compressor, which is assumed to be 70% in this study.

Table 1 – Turbocharger modeling parameters.

Compressor isentropic efficiency (%)	70
Turbine isentropic efficiency (%)	70
Mechanical efficiency (%)	80

By combining Eqns. (7) and (10), it is possible to calculate the compressor work using the isentropic efficiency and PR.

$$W_c = \frac{\dot{m}_c c_p T_{c,in} \left((P_{c,out}/P_{c,in})^{\gamma_c-1/\gamma_c} - 1 \right)}{\eta_{c,s}} \quad (11)$$

For the modeling of the turbine, we follow the similar approaches. The high pressure and temperature exhaust gas from the engine enters the turbine, and is discharged at the hybrid system pressure of 1 bar. During the process, the power generated by the turbine is given by Eqn. (12).

$$\dot{W}_t = \dot{m}_t c_{p,t} (T_{t,in} - T_{t,out}) \quad (12)$$

where $\dot{m}_t$ is the mass flow rate of the gas passing through the turbine, and $T_{t,in}$ and $T_{t,out}$ are the temperatures at the turbine inlet and outlet, respectively.

For an isentropic turbine, the pressure and the temperature ratios of the turbine are given in Eqn. (13).

$$(P_{t,in}/P_{t,out})^{\gamma_t-1/\gamma_t} = \frac{T_{t,in}}{T_{t,out,s}} \quad (13)$$

where $P_{t,in}$ and $P_{t,out}$ are the pressures of the turbine inlet and outlet, respectively, and $T_{t,out,s}$ is the temperature at the turbine outlet by isentropic process.

By using the isentropic efficiency of the turbine in Eqn. (14), the temperature at the non-isentropic turbine outlet is calculated as in Eqn. (15).

$$\eta_{t,s} = \frac{T_{t,in} - T_{t,out}}{T_{t,in} - T_{t,out,s}} \quad (14)$$

$$T_{t,out} = T_{t,in} \left(1 - \eta_{t,s} \left(1 - \left(\frac{P_{t,out}}{P_{t,in}} \right)^{\gamma_t-1/\gamma_t} \right) \right) \quad (15)$$

where the isentropic efficiency of the turbine is assumed to be 70%.

The power output of the turbine can be expressed in Eqn. (16), by combining Eqns. (12) and (15).

$$W_t = \dot{m}_t c_p \eta_{t,s} T_{t,in} \left(1 - \left(\frac{P_{t,out}}{P_{t,in}} \right)^{\gamma_t-1/\gamma_t} \right) \quad (16)$$

Finally, the mechanical efficiency is defined in Eqn. (17), which is to consider the various mechanical losses during the transmission of the power from the turbine to the compressor through the shaft. The value in this study is assumed to be 80%.

$$\eta_m = \frac{W_c}{W_t} \quad (17)$$

3. Engine downsizing

As discussed earlier, engine downsizing is needed to reduce the cost of the system, as well as to reduce the footprint of this additional device to the original fuel cell system. In this section, two strategies are explored to downsize the engine by reducing the volume flow rate in a given mass flow rate from the fuel cell to the engine. In Section 3.1, the use of turbocharger is demonstrated to increase the pressure of the engine

intake charge, and in Section 3.2, the higher geometric compression ratio of the engine is exploited to decrease the required intake charge temperature and thus reduce the volume flow rate.

3.1. Increasing pressure of engine intake charge – turbocharger

In this section, the use of a turbocharger is discussed as one of the methods to downsize the engine in the hybrid system. As mentioned earlier, the higher pressure of the intake charge leads to the lower volume flow rate in a given mass flow rate, and thus the required volume of the engine in the hybrid system can be reduced. A supercharger may also be considered to pressurize the intake charge. However, while the compressor of the turbocharger doesn't require additional power than the one transferred from the companion turbine, supercharger demands extra work from the engine itself and thus there is an efficiency penalty. In this study, we are focusing on the turbocharger application, because the main goal of the hybrid system is to maximize the system efficiency.

In the following sub-sections, the modeling results under various pressure ratios of the turbocharger for the hybrid system (Fig. 1(c)) are presented. The component characteristics, i.e. MCFC stack and HCCI engine, are described first, followed by the discussion on the overall system performances.

3.1.1. Component characteristics under pressure ratio variation – MCFC stack

Table 2 lists the operating conditions for MCFC stack under pressure ratio variation. In this study, the pressure ratio is varied from 1 to 2.25. Since the pressure in the hybrid system, including the inlet pressure of the compressor, is assumed to be 1 bar, these pressure ratios correspond to 1 bar and 2.25 bar at the compressor outlet, respectively. Here, the lower bound represents the naturally-aspirated operation without turbocharger for comparison. The upper bound is rather higher than the typical maximum pressure of ~2 bar (absolute) for the engine application, but this highest number is included to see the potential benefit of the significant boost.

To isolate the effect of MCFC on the system performances under this variation of the turbocharger operation, MCFC anode inlet (state 2) and cathode inlet (state 13) conditions have been kept the same, of which the temperatures are controlled by the amount of heat exchanges in HX1 and HX3, while keeping the maximum effectiveness of the heat exchangers under 85%, as mentioned earlier. The fuel (CH_4) supply rate and the S/C ratio are fixed at 500 kW (LHV basis) and 2.5:1, respectively. In addition, the fuel and air utilization factors are also fixed at 70% and 40%. The reason of this choice for utilization factors is to reduce the concentration over-voltage of the fuel cell operation at the end of the anode and cathode channels by leaving sufficient concentrations of fuel and oxygen. This extra air also prevents the excessive temperature rise in the MCFC by carrying heat out by the outgoing air flow. As a result, MCFC power output and efficiency are almost the same of ~260 kW and ~52.5% throughout the pressure ratio variation, as shown in Table 3.

Table 2 – MCFC operating conditions under pressure ratio variation.

Parameter	PR 1.0	PR 1.25	PR 1.5	PR 1.75	PR 2.0	PR 2.25
LHV CH ₄ (kW)	500					
Fuel utilization factor (%)	70					
Air utilization factor (%)	40					
S/C ratio (H ₂ O: CH ₄)	2.5: 1					
Anode inlet temperature (°C)	571	569	571	571	571	571
Cathode inlet temperature (°C)	581	582	578	578	578	578

3.1.2. Component characteristics under pressure ratio variation – HCCI engine

As discussed in Section 2.3, in an HCCI engine, the combustion timing is largely dependent on the thermodynamic state and the chemical reaction kinetics associated with the reactant mixture. In the present hybrid system, since the engine intake composition is already given by a fuel cell operating condition, the temperature and pressure of the intake charge can be major control variables of the combustion timing, out of which the former has a dominant effect on the chemical reaction rates. Therefore, control on the intake charge temperature is needed to achieve the proper combustion timing, which is also critical for complete combustion. In this study, the combustion timing is represented by the timing of the maximum pressure rise rate (MPRR), because the HCCI combustion occurs for a very short period of time and thus the in-cylinder pressure rises very rapidly.

Fig. 4(a) shows the combustion timings under various pressure ratios tested. They are shown in crank-angle-degree after top dead center (CAD aTDC), where ‘0’ corresponds to the combustion top dead center. It is noted that the combustion timings are kept optimally around 8 CAD aTDC for the reasons discussed in Section 2.3. For completeness of the combustion process, it is shown in Fig. 4(b) that the combustion efficiency is kept higher than 98% in all the tested conditions.

As mentioned above, the combustion timing is a strong function of the engine intake temperature, which is controlled by the amount of heat exchange in HX4 in Fig. 1. Although there is a slight increase in temperature through the compressor, it is still not reaching the required temperature and the additional heating is needed in HX4. Fig. 4(c) shows the resultant engine intake charge temperature for various pressure ratios tested. In order to keep the similar combustion timing as shown in Fig. 4(a), it decreases a little, as the pressure ratio is increased. This is mainly because overall higher pressure of the mixture leads to shorter ignition delay with a given composition, which should be compensated by slightly lowered temperature to keep the similar HCCI combustion timings.

Since the fuel cell system is known for its cleanness in pollutant emissions, it is desirable that the addition of an HCCI engine to the fuel cell system should not cause any additional harmful emissions. Major pollutants considered in

typical internal combustion engines are unburned hydrocarbons (UHC), carbon monoxide (CO), and oxides of nitrogen (NO_x). Among these, the first two components are difficult to be estimated in our zero-dimensional engine modeling, because in-cylinder geometry, especially the crevice volume around the piston, should be considered to accurately estimate it. However, for the production of oxides of nitrogen, the peak temperature during combustion process is known to be the most determining factor, and the production rate exponentially increases above ~1800 K. Fig. 4(d) shows the peak temperatures during the engine cycle for the tested conditions, and they are all around 1700 K, much under 1800 K. This low peak temperature is due to the diluted nature of the anode off-gas where only small amount of fuel chemical energy is left for the combustion in the HCCI engine. Therefore, it is expected that there is almost no NO_x in the exhaust gas of the HCCI engine for the hybrid system operation.

Fig. 4(e) shows peak pressure during the combustion process. The peak pressure increases almost linearly as the pressure ratio of the compressor rises. Since the engine geometric compression ratio is fixed at 12.5 and the reactant mixtures contain similar amount of fuel chemical energy for exothermicity during the HCCI combustion, the peak pressure increase can be explained mainly by the higher intake charge pressure from the compressor. At pressure ratio of 2.25, the peak pressure in the engine is already near 70 bar, which may be detrimental to the mechanical parts during the real engine operation, and thus the compression above this point may not be desirable.

In Fig. 4(f), engine power output and efficiency are depicted for the various pressure ratios. Nearly constant power and thus similar efficiency, is produced from the engine despite the relatively large variation of the intake charge pressure. The slight decrease with higher pressure ratio operation is mainly due to the increased pumping loss with the higher exhaust back pressure caused by the turbocharger.

In order to show the potential of downsizing by using the turbocharger, Fig. 5 depicts the downsizing ratio in the engine displacement volume, where the 100% point corresponds to 19.2 L of the HCCI engine without turbocharger or at pressure ratio of unity. Downsizing by a turbocharger proves itself to be an effective way, in that the pressure ratio of 1.25 already achieves 75% engine downsizing. The engine size falls around

Table 3 – MCFC performances under pressure ratio variation.

Parameter	PR 1.0	PR 1.25	PR 1.5	PR 1.75	PR 2.0	PR 2.25
Net MCFC Output (kW)	263	262	263	263	263	263
MCFC Efficiency (%)	52.6	52.4	52.5	52.6	52.6	52.5

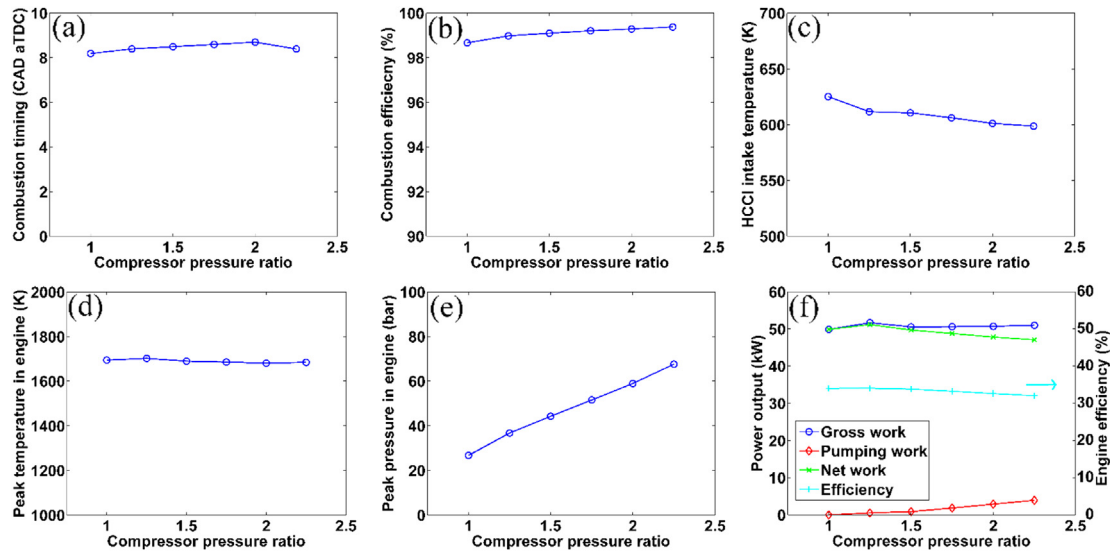


Fig. 4 – HCCI engine under pressure ratio variation: (a) combustion timing, (b) combustion efficiency, (c) intake charge temperature, (d) peak temperature, (e) peak pressure, and (f) power output and efficiency.

40% at the highest pressure ratio tested, which implies that the final engine size is ~ 7.6 L.

3.1.3. System performance

From Sections 3.1.1 and 3.1.2, it is already expected that the hybrid system performance is little affected by the turbo-charger operation. Fig. 6 demonstrates nearly constant power output and efficiency of the hybrid system as a whole, as the compressor pressure ratio varies. There is a little decrease with the higher pressure ratio, which is mainly due to the increased pumping work by turbocharger in the HCCI engine operation, as mentioned earlier.

In summary, a turbocharger was successfully applied to the hybrid system without affecting overall system performances, while maximizing the potential of engine downsizing. Still, the peak pressure during the engine operation should be considered when determining the maximum pressure ratio. Additionally, off-design operation with this additional degree of freedom by the turbocharger should be carefully examined. This last point will be discussed in Section 4.

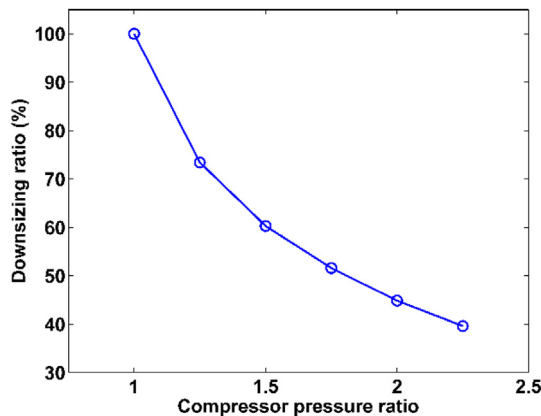


Fig. 5 – HCCI engine downsizing ratio under pressure ratio variation.

3.2. Decreasing temperature of engine intake charge – high geometric compression ratio

In this section, we use the same hybrid system shown in Fig. 1(c), with which new engine design is suggested to reduce the intake charge temperature requirement, while keeping proper HCCI combustion timing and combustion efficiency. If the temperature of the gas is decreased, the volume flow rate for a given mass flow is also decreased due to increased density, thus leading to smaller engine volume requirement. When the temperature of the intake charge in a given composition is lowered, the ignition delay becomes longer and, at certain minimum intake temperature condition, the HCCI combustion timing is too much delayed into the expansion stroke, leading to incomplete combustion or even misfire. To remedy it, the increase in geometric compression ratio of the HCCI engine is suggested. At higher compression ratio, the mixture can achieve comparable post-compression temperature with lower intake temperature than otherwise. It is noted that the geometric compression ratio is not a variable that can

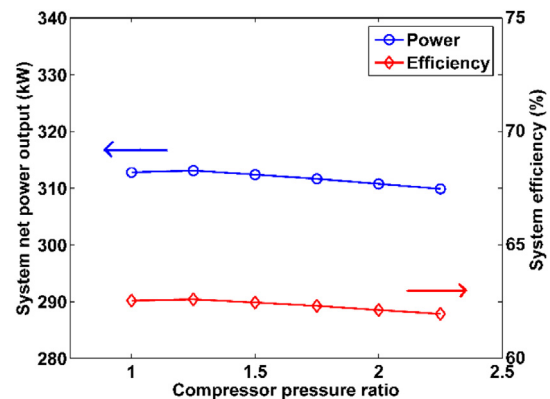


Fig. 6 – Net power output and efficiency of hybrid system under pressure ratio variation.

be changed during the operation, but should be designed and implemented when the engine is manufactured.

In the following sub-sections, the modeling results with different geometric compression ratio of the HCCI engine in the hybrid system are described. The component characteristics of MCFC stack and HCCI engine are presented first, and the overall hybrid system performances are shown in the end. In order to understand the potential of downsizing by use of high geometric compression ratio of the engine only, the pressure ratio of the turbocharger is fixed at unity.

3.2.1. Component characteristics under compression ratio variation – MCFC

The geometric compression ratio (CR) of the engine in our original hybrid system was 12.5, which was chosen to match the typical compression ratio of the gasoline-fueled HCCI engine. In this study, for downsizing purpose, the geometric compression ratio is increased up to 20, which corresponds to the typical upper limit of the conventional diesel-fueled compression ignition engine.

The operating conditions for the MCFC stack under the geometric compression ratio variation are shown in Table 4. To isolate the effect of the fuel cell under the variation, the amount of fuel entering the fuel cell, fuel and air utilization factors, and steam-to-carbon ratio are fixed for all the tested conditions. In addition, the temperatures for anode inlet (state 2) and cathode inlet (state 13) are kept almost the same. As a result, the MCFC stack can produce almost constant power of ~263 kW and constant efficiency of ~53% with all these different HCCI engines of various compression ratios, as shown in Table 5.

3.2.2. Component characteristics under compression ratio variation – HCCI engine

Fig. 7(a) shows the engine intake charge temperature and post-compression temperature at the top dead center under various geometric compression ratios. As mentioned in the preceding section, intake temperature can be lowered almost linearly with the compression ratio, while achieving the similar level of post-compression temperature. These intake temperatures are selected to keep the combustion timing near 8 CAD aTDC, which ensures the combustion efficiency over 98%, as shown in Fig. 7(b).

Fig. 7(c) shows peak temperature during combustion. Since both the post-compression temperature and the heating values of the reactant mixture are at similar level among compression ratios, it is already expected that the peak temperatures should be similar. In addition, they are all under 1700 K, thus little NO_x emissions are expected.

Table 4 – MCFC operating conditions under compression ratio variation.

Parameter	CR 12.5	CR 15	CR17.5	CR 20
LHV CH ₄ (kW)	500			
Fuel utilization factor (%)	70			
Air utilization factor (%)	40			
S/C ratio (H ₂ O : CH ₄)	2.5: 1			
Anode inlet temperature (°C)	571	571	571	571
Cathode inlet temperature (°C)	581	579	579	578

Table 5 – MCFC performances under compression ratio variation.

Parameter	CR 12.5	CR 15	CR17.5	CR 20
Net MCFC Output (kW)	263	263	263	263
MCFC Efficiency (%)	52.6	52.6	52.6	52.6

As mentioned earlier, the intake charge pressure is kept at 1 bar, and thus the higher geometric compression ratio leads to higher post-compression pressure and, finally, higher peak pressure during combustion, as shown in Fig. 7(d). However, even under the highest compression ratio of 20, it is below 50 bar, which wouldn't cause any significant material stresses.

As shown in Fig. 7(e), the higher compression ratio operation leads to the increase in the engine efficiency. Since the reactant mixtures have almost the same level of heating values for all the tested conditions, this higher efficiency also results in higher power output. It is generally true that the use of high compression ratio can increase the engine efficiency, but above a certain level, it may rather cause efficiency degradation due to rapid increase in heat transfer with higher post-compression and combustion temperatures. However, in our strategy, post-compression and peak temperatures are kept similar under various compression ratios, as noted in Fig. 7(a) and (c), with which the modified Woschni heat transfer model predicts little difference in heat transfer for all the tested conditions, as shown in Fig. 7(f).

Finally, Fig. 8 shows the potential of the engine downsizing by use of the increased geometric compression ratio. With the displacement volume of the engine at CR of 12.5 as the reference size (19.2 L), the highest CR tested of 20 can lead to the engine displacement volume of ~16.9 L which is ~12% reduction. This is relatively small downsizing potential as compared to the one by the use of turbocharger for the tested range of operations.

3.2.3. System performance

Since the power output and efficiency of the MCFC stack are nearly constant (in Table 5) while those of the HCCI engine gradually increase with higher compression ratio (in Fig. 7(e)), the hybrid system power output and efficiency are slightly increased, as shown in Fig. 9.

In summary, the proper selection of the engine geometric compression ratio can be a way to downsize the engine in the suggested hybrid system. However, the downsizing potential under reasonable CR variation is somewhat limited, as compared to the use of turbocharger in Section 3.1. There is a small additional benefit with high CR operation that the HCCI engine efficiency and power output increases a bit in our simulation results, but the possible penalty on the high peak pressure and the increased heat transfer should be considered, when the engine is built for practice.

4. Off-design operation

In this section, the strategy to operate the hybrid system at off-design conditions is discussed. In general, a power-generating system runs at constant power output for most

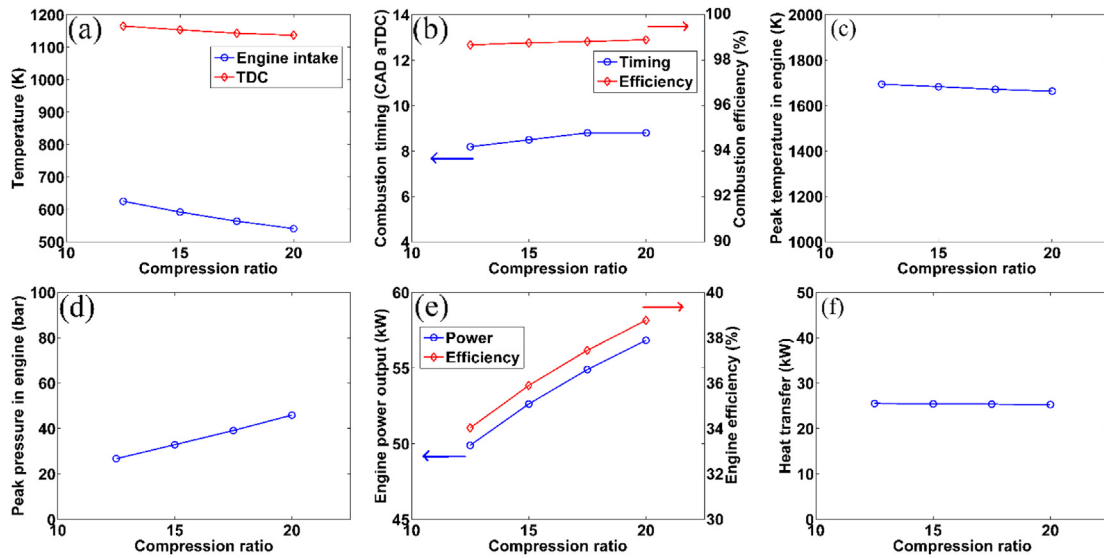


Fig. 7 – HCCI engine under compression ratio variation: (a) intake charge temperature and post-compression temperature at TDC, (b) combustion timing and efficiency, (c) peak temperature, (d) peak pressure, (e) power output and efficiency, and (f) heat transfer.

of its life-time, but it still needs to operate at various power levels under circumstances. Therefore, in developing a new hybrid system, it is important to demonstrate its capability to cover these various operating conditions. In the following subsections, we firstly introduce the hybrid system with optimally-downsized engine based on the analyses from Section 3 (Section 4.1), and the strategy (Section 4.2) and the performance (Section 4.3) of the newly-developed system under power variation are discussed.

4.1. Optimized MCFC-HCCI engine hybrid system at design point

In Section 3, the two methods for downsizing the HCCI engine, i.e. the use of turbocharger and high geometric compression ratio, were discussed. In this section, the new hybrid system is proposed by combining these two strategies. It is reminded that the turbocharger was analyzed under various compressor

pressure ratios, from 1 to 2.25, and the geometric compression ratio of the HCCI engine was varied between 12.5 and 20. Although both higher compressor pressure ratio and geometric compression ratio can be combined to further increase the downsizing potential, there is a possibility that the engine may suffer mechanical issues by resultant high peak pressures in a real-world application. For this reason, we conservatively choose 1.5 and 15 for the pressure ratio of the turbocharger and the geometric compression ratio of the engine, respectively, both of which are slightly under the median values tested. This choice of parameters result in the moderate peak pressure (under 60 bar) and peak temperature (around 1700 K) at design point, as shown in Fig. 10. The downsized engine specification and operating conditions at design point are listed in Table 6. The engine volume is already ~60% level of the one in the original hybrid system.

The MCFC stack operating conditions and the stream properties for the newly-developed hybrid system at design point are shown in Tables 7 and 8, respectively. The state

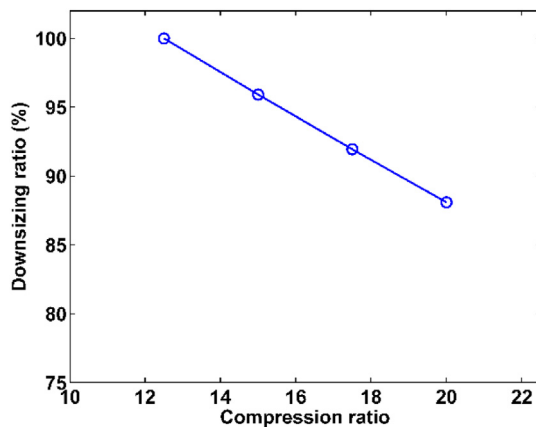


Fig. 8 – HCCI engine downsizing ratio under compression ratio variation.

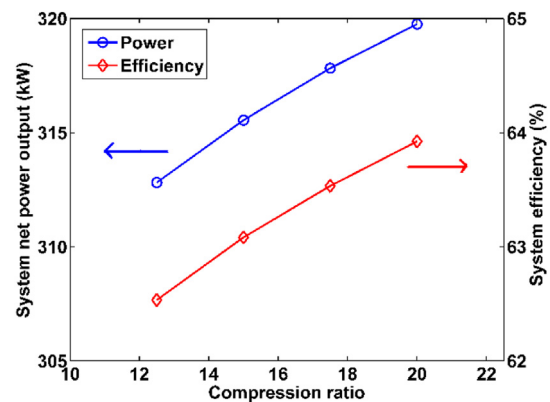


Fig. 9 – Net power output and efficiency of hybrid system under compression ratio variation.

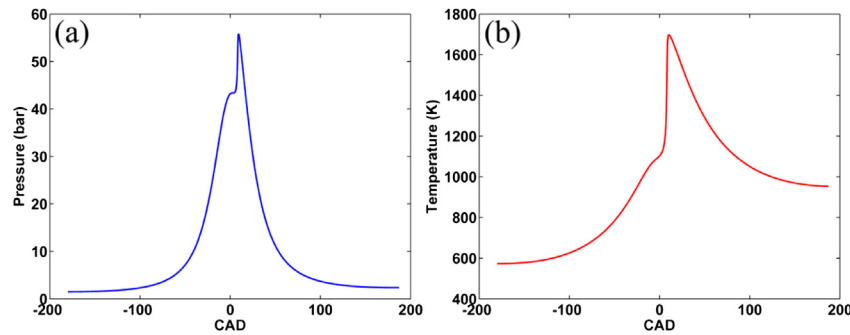


Fig. 10 – HCCI engine at design point: (a) in-cylinder pressure and (b) in-cylinder temperature profiles.

numberings in Table 8 are in line with those in Fig. 1. The same inlet thermodynamic states and utilization factors as before are used for the MCFC operation. By integrating the turbocharger into the system, there are additional states to be noted. The air-fuel mixture (state 6a) enters the compressor at 1 bar, 58 °C, and then the mixture (state 7) exits the compressor at 1.5 bar, 107 °C. The engine exhaust gas (state 8a) at 1.59 bar and 610 °C, which enters the turbine, expands to the atmosphere pressure gas (state 9) at 1 bar and 561 °C. The system performance at design point, e.g. power output and efficiency, will be discussed with those under off-design operations in Section 4.3.

4.2. Operating strategy for power variation

In this section, the optimized hybrid system developed in Section 4.1 is analyzed at off-design operations with various fuel cell power output levels, especially at part loads. A part-load operation for both standalone and hybrid fuel cell systems has been studied by several researches [6,12,13,20]. Liu et al. investigated MCFC and micro-GT hybrid system at part-load operation by using various control methods, which were varying fuel quantity, changing the speed of gas turbine, and using air bypass setup [6]. Sanchez et al. showed MCFC-Stirling engine hybrid system operation at part load by controlling the fuel mass flow and utilization factor [12]. Escalona et al. analyzed several hybrid systems at part-load operations. The bottoming cycles of their hybrid systems were configured either by SCO_2 turbine or Stirling engine, and the hybrid systems were operated under various load range by controlling the shaft speed of the turbine or the engine [13]. Marra et al.

simulated 1-MW standalone MCFC system for 25–100% load operation, and compared the performance results [20].

In this section, the MCFC-HCCI engine hybrid system is analyzed at 65–100% fueling levels of the design point to demonstrate a possibility that the hybrid system can be driven under various power output requirements. Here, the percentage of the fueling level is based on the methane flow rate (while keeping the same S/C ratio) into the hybrid system. The reason for choosing the “fueling” level as the test parameter, instead of the “power output” level, is simply for convenience purpose in performing simulation. However, it will be shown that the power output level is very close to the fueling level, due to the similar system efficiencies under the tested conditions.

Since the proposed hybrid system has two independently operable components, i.e. MCFC stack and HCCI engine, coupled together, it requires systematic control strategies for off-design operation. Fig. 11(a) depicts fuel and air flow rates variation under part-load operations. Firstly, fuel mass flow rate (state 1) is reduced to the desired fueling level, while keeping the same fuel utilization at 70% in the MCFC stack. Secondly, air entering the cathode of MCFC (state 10) is also reduced in the same rate as the decreased fueling rate, which is to keep the air utilization near 40%. Thirdly, since the mass flow rate of the anode off-gas and thus the fuel components, i.e. CO and H_2 , decrease with lowered fueling level, the mass flow rate of the air entering the HCCI engine (state st) should also be decreased to keep the stoichiometric proportion between fuel and oxidizer in the intake mixture to the HCCI engine. The advantage of using the stoichiometric mixture for the engine operation is the possible use of an ad-hoc three-way catalytic converter as an after-treatment device for the HCCI engine, which is to completely remove any harmful components to be fed into the fuel cell stack. We consider it as

Table 6 – Engine geometry and operating conditions at design point.

Bore (mm)	121
Stroke (mm)	121
Cylinders	8
Displacement (L)	11.1
Pressure ratio	1.5
Compression ratio	15
Engine speed (rpm)	1800
Equivalence ratio	1 (Stoichiometric)

Table 7 – MCFC stack operating conditions at design point.

LHV CH_4 (kW)	500
Fuel utilization factor (%)	70
Air utilization factor (%)	40
S/C ratio ($\text{H}_2\text{O}:\text{CH}_4$)	2.5: 1
Anode inlet temperature (°C)	570
Cathode inlet temperature (°C)	582

Table 8 – Stream properties for hybrid system at design point.

State	T(°C)	P(bar)	$m_{\text{dot}}(\text{kg s}^{-1})$	Mass concentration (%)							
				Ar	CH ₄	CO	CO ₂	H ₂	H ₂ O	N ₂	O ₂
1	30	1	0.0381	—	26.3	—	—	—	73.7	—	—
1a	280	1	0.0381	—	26.3	—	—	—	73.7	—	—
2	570	1	0.0381	—	26.3	—	—	—	73.7	—	—
3	658	1	0.1524	—	—	3.82	67.0	0.50	28.7	—	—
4	541	1	0.1524	—	—	3.82	67.0	0.50	28.7	—	—
5	110	1	0.1524	—	—	3.82	67.0	0.50	28.7	—	—
5a	65	1	0.1524	—	—	3.82	67.0	0.50	28.7	—	—
6	65	1	0.1324	—	—	4.40	77.1	0.58	17.9	—	—
6a	58	1	0.1324	—	—	3.37	59.1	0.44	13.7	17.9	5.45
7	107	1.50	0.1728	—	—	3.37	59.1	0.44	13.7	17.9	5.45
8	301	1.50	0.1728	—	—	3.37	59.1	0.44	13.7	17.9	5.45
8a	610	1.59	0.1728	—	—	3.37	59.1	0.44	13.7	17.9	5.45
9	561	1	0.1728	—	—	0.05	64.3	—	17.7	17.9	0.04
10	30	1	0.3271	—	—	—	—	—	—	76.7	23.3
11	305	1	0.3271	—	—	—	—	—	—	76.7	23.3
12	595	1	0.3271	—	—	—	—	—	—	76.7	23.3
13	582	1	0.4999	—	—	—	22.2	—	6.11	56.4	15.3
14	658	1	0.3856	—	—	—	7.08	—	7.92	73.1	11.9
15	436	1	0.3856	—	—	—	7.08	—	7.92	73.1	11.9
15a	391	1	0.3856	—	—	—	7.08	—	7.92	73.1	11.9
16	278	1	0.3856	—	—	—	7.08	—	7.92	73.1	11.9
st	30	1	0.0404	—	—	—	—	—	—	76.7	23.3

the most efficient and cheapest solution for the real-world application. Finally, since the mass flow rate of intake mixture to the HCCI engine decreases at part load, to keep the same volume flow rate as in the design point, the intake pressure should be decreased. This can be achieved by lowering the compressor pressure ratio in the turbocharger operation, as shown Fig. 11(b). We might also increase the intake temperature for higher volume flow rate, but, as discussed earlier, it would change the combustion timing significantly, which is to be avoided. It is noted that the combustion timings are kept near 8 CAD aTDC for all the part load conditions tested.

4.3. System performances

As the fueling level decreases, net power output of the system also decreases almost in the same proportion, while the overall system efficiency increases a little, as depicted in Fig. 12.

Table 9 lists the detailed performance values of MCFC stack and HCCI engine under part-load operations. When the power

output requirement is lowered, MCFC operates at lower current density, which leads to slightly increased efficiency due to reduced fuel cell losses. For the HCCI engine, however, the efficiency is nearly the same at all power levels tested, due to the similar intake thermodynamic states (except pressure) and combustion timings.

In summary, the proposed hybrid system of MCFC and HCCI engine is demonstrated to operate at part loads down to 65% of the designed power level. The fuel and air feeding rates and the boost level in the turbocharger should be controlled to keep each component working properly. In the end, the system efficiency is kept nearly the same at all the tested conditions.

5. Conclusions

We previously developed a new hybrid system of MCFC and HCCI engine, which demonstrated increased power output and efficiency, as compared to a standalone MCFC counterpart. In this study, improvement on the HCCI engine design in

Table 9 – Hybrid system performances under power variation.

Parameter	Hybrid system (100%)	Hybrid system (95%)	Hybrid system (90%)	Hybrid system (85%)	Hybrid system (80%)	Hybrid system (75%)	Hybrid system (70%)	Hybrid system (65%)
LHV CH ₄ (kW)	500	475	450	425	400	375	350	325
Utilization factor (%)	70							
Net MCFC Output (kW)	262	250	237	226	213	200	187	175
MCFC Efficiency (%)	52.3	52.6	52.6	53.1	53.3	53.4	53.5	53.9
Engine Output (kW)	53.6	50.7	49.4	44.9	42.2	40.0	37.4	33.7
Engine Efficiency (%)	35.6	35.6	35.9	35.5	35.5	35.7	35.6	35.4
Net Power Output (kW)	315	301	286	271	256	240	225	209
System Efficiency (%)	63.0	63.2	63.6	63.7	63.9	64.1	64.2	64.3

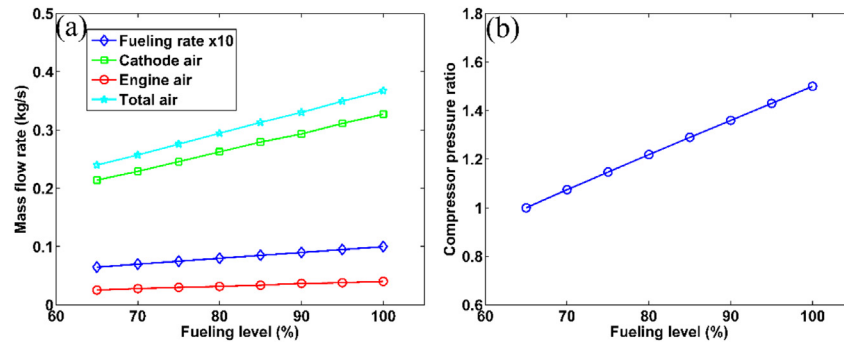


Fig. 11 – (a) Intake mass flow rates for hybrid system and (b) compressor pressure ratio under power variation.

the hybrid system and development of system operating strategies under part-load conditions were carried out.

In the new engine design, we were focusing on downsizing of the HCCI engine to reduce the system cost and the footprint of this additional device in the overall system. Two strategies were suggested, i.e. the use of a turbocharger and the high geometric compression ratio of the engine. To demonstrate the downsizing potential, parametric studies were performed for various pressure ratios of the turbocharger and for range of geometric compression ratios. Secondly, based on the results of these parametric studies, a new engine design was chosen for the hybrid system, with which the operating strategy for power variation at part loads was developed for its practicality in real-world application.

Here are the summary of the findings:

1. The use of a turbocharger proves to be an effective way of engine downsizing. For changing the pressure ratio (PR) of the compressor from 1 to 2.25, it is predicted that the engine size can be decreased to the level of 40% at the highest PR tested, as compared to that of non-boosting condition. Even with this improvement in engine downsizing, the hybrid system power output and efficiency are little affected.
2. Changing the compression ratio can be used for engine downsizing, although the effect is relatively limited, as compared to the turbocharger system, in a tested range of

operation. At the compression ratio of 12.5, the engine size is 19.2 L, while at 20, the engine size is reduced to 16.9 L, which is ~12% reduction. The hybrid system power output and efficiency increases a little, mainly due to higher efficiency from the engine operation. At compression ratio of 20, the net power output of the hybrid system is 320 kW and the efficiency 63.9%.

3. To develop the operating strategy for power variation, the new HCCI engine for the hybrid system is chosen with the geometric compression ratio of 15 and the turbocharger operating at pressure ratio of 1.5 at design point. This choice allows the engine downsizing by ~60%, as compared to the previously developed system, without significant stress on the mechanical parts of the engine by limiting the peak pressure under 60 bar. With this new engine design, the hybrid system efficiency already gains ~0.5% at design point. To operate the newly-designed system at varying power levels, fuel and air flow rates are adjusted, and the proper turbocharger operating strategies are suggested. It is demonstrated that the system can run successfully at part-load operations (down to 65% of the fueling level at design point) with the similar hybrid system efficiencies.

Acknowledgments

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Appendix A

In this appendix, modeling process of an HCCI engine is discussed in details. The original contents are from the author's previous work [15] and some of the major equations are reproduced here.

The present model assumes a homogeneous thermodynamic state for in-cylinder mixture and only includes compression and expansion strokes without gas exchange. A

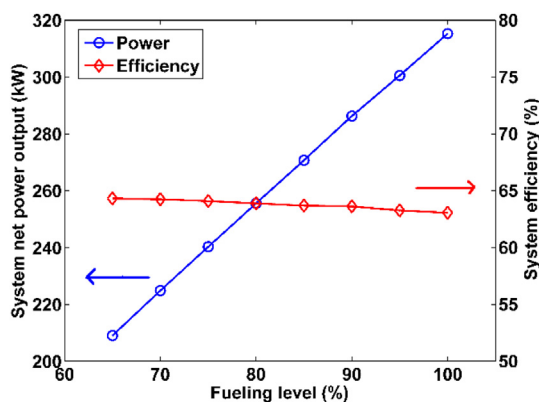


Fig. 12 – Hybrid system power output and efficiency under power variation.

series of ordinary differential equations are solved for the state variables—crank angle, cylinder volume, temperature, and mass of species—by using an ode solver provided by the Mathworks MATLAB.

The engine is running at a constant speed of n RPM, and the piston location is given by crank angle θ

$$\dot{\theta} = n \quad (\text{A.1})$$

The instantaneous cylinder volume (V) is updated by a slider–crank relationship from Heywood [21]

$$\dot{V} = \frac{\pi B^2 a \dot{\theta}}{4} \left(\sin \theta + \frac{a \sin \theta \cos \theta}{\sqrt{l^2 - a^2 \sin^2 \theta}} \right) \quad (\text{A.2})$$

where B is the bore, a is the crank radius, and l is the connecting rod length. The energy equation is solved for the instantaneous cylinder temperature T . By using the first law of thermodynamics for a closed, reacting system, it is possible to write

$$\dot{U} = -P\dot{V} - \dot{Q} \quad (\text{A.3})$$

$$mC_v \dot{T} + \sum_i \dot{m}_i u_i + P\dot{V} + \dot{Q} = 0 \quad (\text{A.4})$$

$$\rightarrow \dot{T} = -\frac{1}{mC_v} \left(\sum_i \dot{m}_i u_i + P\dot{V} + \dot{Q} \right) \quad (\text{A.5})$$

where U is the total (extensive) internal energy, m is the total mixture mass, C_v is the specific heat at constant volume for the total mixture, m_i and u_i are the mass and internal energy of the i -th species, respectively, P is the instantaneous cylinder pressure, Q is the heat transfer amount. Here, the production (or destruction) rate of species mass m_i by chemical reactions is evaluated by using a function from the Cantera toolbox, which refers to GRI-Mech 3.0 mechanisms. Finally, the instantaneous cylinder pressure P is evaluated by using the ideal gas law

$$P = \frac{mRT}{V} \quad (\text{A.6})$$

where R is the mass-based gas constant.

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